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A highly efficient six-stroke internal combustion engine cycle with water injection for in-cylinder exhaust heat recovery

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ABSTRACT

A concept adding two strokes to the Otto or Diesel engine cycle to increase fuel efficiency is presented here. It can be thought of as a four-stroke Otto or Diesel cycle followed by a two-stroke heat recovery steam cycle. A partial exhaust event coupled with water injection adds an additional power stroke. Waste heat from two sources is effectively converted into usable work: engine coolant and exhaust gas. An ideal thermodynamics model of the exhaust gas compression, water injection and expansion was used to investigate this modification. By changing the exhaust valve closing timing during the exhaust stroke, the optimum amount of exhaust can be recompressed, maximizing the net mean effective pressure of the steam expansion stroke (MEP_{steam}). The valve closing timing for maximum MEP_{steam} is limited by either 1 bar or the dew point temperature of the expansion gas/moisture mixture when the exhaust valve opens. The range of MEP_{steam} calculated for the geometry of a conventional gasoline engine and is from 0.75 to 2.5 bars. Typical combustion mean effective pressures ($MEP_{combustion}$) of naturally aspirated gasoline engines are up to 10 bar, thus this concept has the potential to significantly increase the engine efficiency and fuel economy.

1. Introduction

In internal combustion engines, a significant amount of the fuel energy exits the engine in the form of thermal energy in the exhaust. To demonstrate this, Fig. 1 shows previously unpublished experimental data from a turbo-charged 2007 Saab Biopower vehicle during a federal test protocol (FTP)-75 engine cycle, the engine cycle that is largely used to calculate the EPA "city" fuel economy. The FTP-75 cycle is highly transient with numerous stops and starts, and corresponding fluctuations in the engine-out exhaust temperature from 400 to 600 °C. The exhaust temperature range of naturally aspirated gasoline engine is higher, typically from 450 to 800 °C. The total fuel energy consumed during the course of the driving cycle is approximately 58.5 MJ, or about 1.7 L (0.45 gallons) of unleaded gasoline fuel. The percentage of fuel energy converted to useful work for this driving cycle (i.e. the vehicle thermal efficiency) is 10.4%. A much larger portion of fuel energy, 27.7%, exits the vehicle in the form of thermal energy in the exhaust, while the remaining 61.9% of the energy balance consists of energy losses to friction, coolant, and other. For these series of tests, the instrumentation was limited to the fuel and exhaust systems, and thus the friction, coolant, and other losses are indistinguishable. Only a portion of the energy in the exhaust is available for recovery due to process irreversibilities, ambient conditions, etc. The exergy in the exhaust was determined by a 2nd law thermodynamic analysis as described by Rodriguez [1]. The difference in brake work fractions between the 1st and 2nd law analyses, also shown in Fig. 1, is because the 1st law analysis of energy distribution is normalized with the lower heating value of the fuel while the 2nd law analysis is normalized with the exergy of the fuel. The 2nd law exhaust exergy of 8.4% as shown in Fig. 1, is nearly as high as the amount of brake work. Thus, there is an abundant amount of available energy present in the exhaust of modern gasoline vehicles that can be used to improve overall system efficiency if an effective means of energy recovery can be employed. The concept posed here will utilize a portion of this previously wasted energy in the exhaust, but also will use a portion of the coolant waste energy to provide additional shaft work that would otherwise be discarded.

Improving the efficiency of internal combustion engines is an ongoing area of active research. Numerous designs have been proposed based on the traditional Otto or Diesel cycles, and all of these include four sequential thermodynamic processes or 'strokes' of the piston. These are the following strokes: 1) air–fuel intake, 2)

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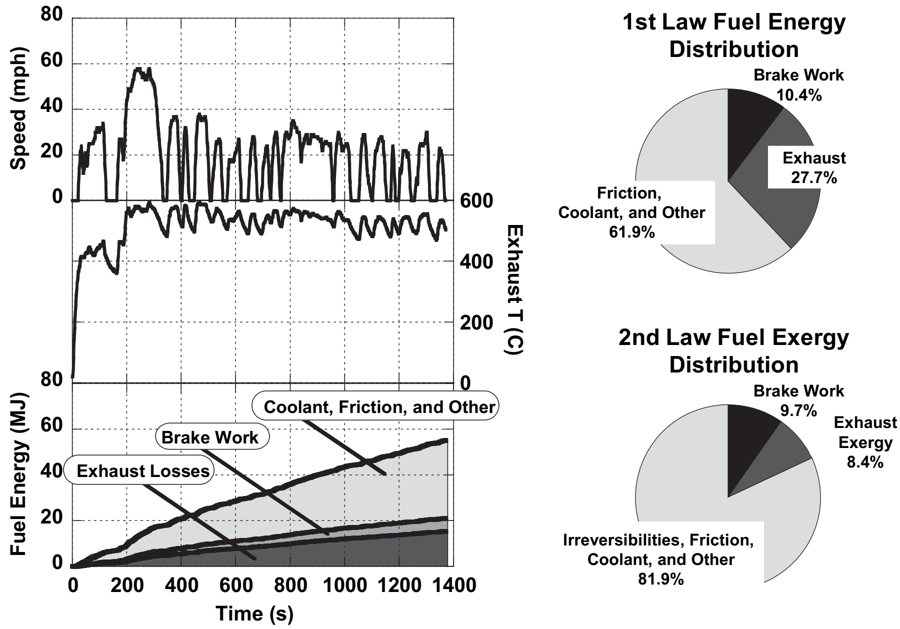

Fig. 1. Federal test protocol (FTP) test cycle for a 2007 Saab Biopower showing speed, exhaust temperature, consumed fuel energy, and recoverable exhaust energy. Experimental data were collected at the ORNL chassis dynamometer facility.

air–fuel compression, 3) post-combustion expansion, and 4) exhaust gas discharge. Fig. 2 illustrates a schematic of the typical four-stroke sequence for an Otto cycle, and Fig. 3 illustrates the corresponding pressure–volume trace.

The modified cycle proposed here adds two additional strokes that increase the work extracted per unit input of fuel energy. These additional strokes involve trapping and recompression of some of the exhaust from the fourth piston stroke, followed by a water injection and expansion of the resulting steam/exhaust mixture. The residual exhaust gas is trapped in the cylinder by closing the exhaust valve earlier than usual, i.e., well before top center (TC). Energy from the trapped recompressed exhaust gases is transferred to the liquid water, causing it to vaporize and increase the pressure. This added pressure then produces more work from another expansion process. The steam–exhaust gas mixture is expelled to ambient pressure near the point of maximum expansion. The modified sequence of strokes is illustrated in Fig. 4, and the corresponding pressure–volume trace is shown in Fig. 5.

To summarize in graphical form on Fig. 6, representative valve lifts and resultant representative combustion chamber pressure traces are superimposed versus crank angle where the proposed exhaust recompression and water injection are explicitly shown.

Thus, the additional proposed power stroke produces more output work from the engine without any additional fuel, thereby increasing the fuel efficiency of the engine.

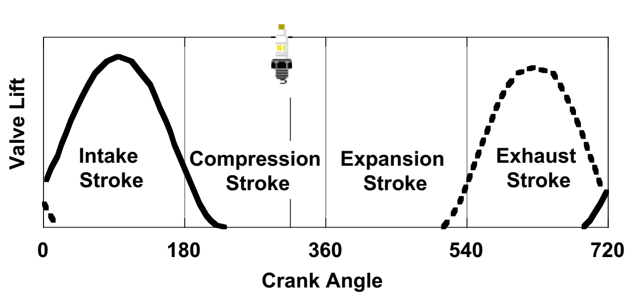

Fig. 2. Schematic of typical intake (—) and exhaust (- - -) valve events for a gasoline engine.

A six-stroke engine cycle that couples a two-stroke steam cycle after a four-stroke Otto or Diesel cycle is not entirely new, and several patents have been awarded for similar concepts. The first of these was awarded to Dyer in 1920 [2], followed by numerous variations on very similar cycles with more detailed descriptions of components and recovery loops [3–8]. The engine cycles described in Refs. [2–7] all utilize a complete exhaust stroke during crank angle (CA) 540–720, so that when water is injected it impinges on the combustion chamber surfaces. In addition, a complete exhaust stroke is the preferred configuration in Ref. [8], although an alternate method of recompressing the exhaust gas is mentioned. Thus, the primary sources of heat to vaporize the water in these engine cycles are the combustion chamber surfaces rather than the exhaust gases. The engine cycle described here, which uses water injection to absorb the heat from the exhaust gas directly, is a more

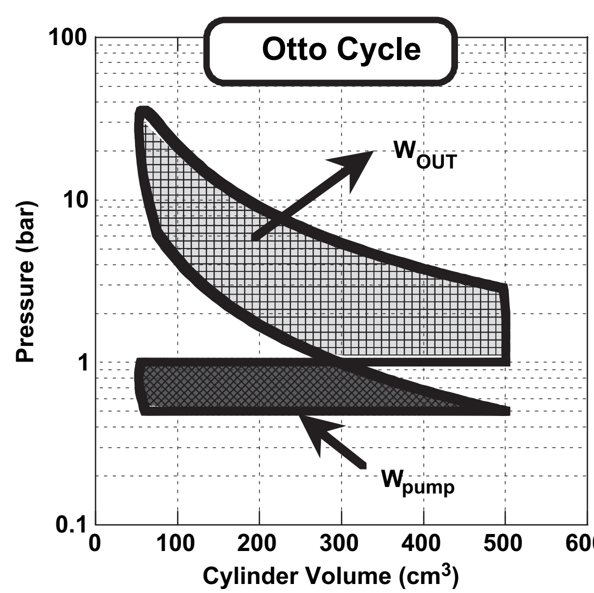

Fig. 3. Schematic of pressure vs. volume for a typical gasoline engine Otto cycle.

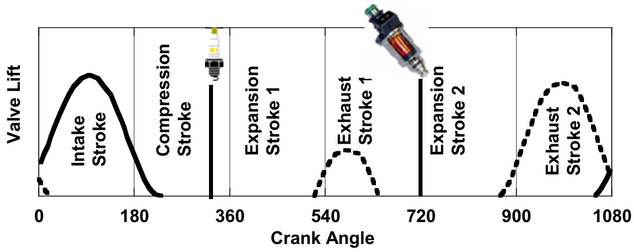

Fig. 4. Schematic of typical intake (—) and exhaust valve events (---) for the six-stroke engine cycle.

practical cycle because ideally there is no impingement of the water on the combustion chamber surfaces, which reduces the likelihood of material incompatibilities.

2. Ideal thermodynamic analysis of the additional power stroke

The analysis starts with the First Law of Thermodynamics for a control volume, given as Energy in − Energy out = Change in stored energy. For any component process of the engine cycle, this is written as [9]

$$Q_{\rm in} - Q_{\rm out} + W_{\rm in} - W_{\rm out} + m_{\rm in}h_{\rm in} - m_{\rm out}h_{\rm out} = \Delta E \tag{1}$$

Eq. (1) holds for every stroke in the cycle, but we will focus here on the exhaust recompression and the additional power stroke from the water injection, CA 540–900. For the purposes of this analysis, subscript 1 will denote the state when the exhaust valve is closed early, subscript 2 represents the initial state at the minimum cylinder volume (clearance volume, or top center TC) immediately prior to water injection, subscript 3 denotes the final state just after the water injection at TC, and subscript 4 denotes the thermodynamic state when the exhaust valve is opened to discard the spent combustion products/water vapor mixture. The exhaust valve opening for State 4 is assumed to occur at the maximum volume of the engine cylinder, or bottom center (BC). The engine state points are graphically depicted in Fig. 7. The convention of upper case denoting extensive and lower case denoting intensive properties is

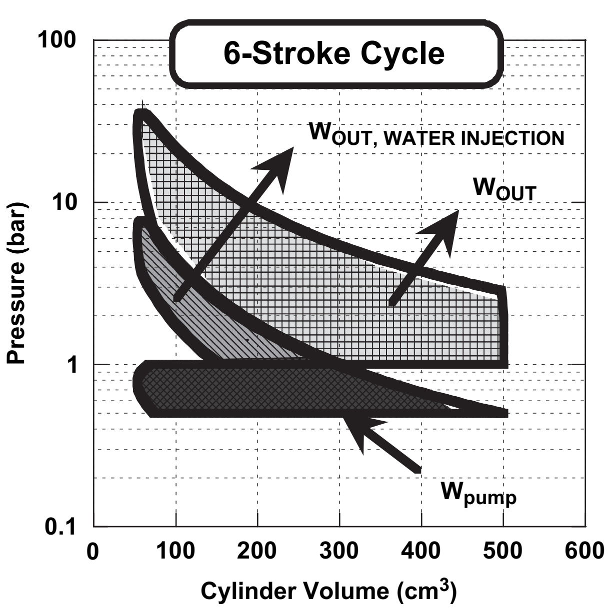

Fig. 5. Schematic of pressure vs. volume for a six-stroke engine cycle.

observed here. In addition, all energy associated with fluid movement or body forces is neglected, as is friction. The thermodynamic control volume used here is the combustion chamber.

A computer program was written to solve the energy equation Eq. (1) subject to the appropriate initial conditions and assumptions described as follows for the cycle component processes. The state properties of the water and the water/exhaust gas mixture were calculated using FluidProp [10].

2.1. Recompression

First, the energy conservation equation for the additional compression of the combustion gas from the time of the early exhaust valve closing (State 1) to the end of compression at TC (State 2) is analyzed and simplified. As yet unspecified, the actual crank angle (CA) for the early exhaust valve closing will be a parameter for investigation. The temperature and pressure at State 1 will also be specified using appropriate values. Because there will be no mass flow out of the combustion chamber control volume during the recompression process, and also assuming that the recompression process is adiabatic, Eq. (1) reduces to

$$W_{\rm in} = E_2 - E_1 \tag{2}$$

Because the temperature and pressure are specified at the exhaust valve early closing, state point 1 is completely determined. Specifying the crank angle when the exhaust valve is closed as a parameter yields the combustion chamber volume at that point, thus determining the mass of combustion gas for States 1 and 2. The thermodynamic state at point 2 is only partially determined with the specific volume being known at TC. An additional assumption that the recompression process is isentropic from State 1 to State 2 yields the additional state property required by the State Postulate of Thermodynamics for a simple compressible system [9] to determine completely the thermodynamic properties at State 2. The work required by the recompression process is thus known for a given crank angle closing.

2.2. Water injection

Now with state point 2 (immediately before water injection) determined, the simple control volume for analysis reduces to a fixed volume at TC with an as yet unspecified entering mass of water with a given enthalpy or temperature. Assuming adiabatic conditions and that the water injection is instantaneous, the energy conservation equation reduces to

$$m_{\rm water}h_{\rm water} = m_3u_3 - m_2u_2 \tag{3}$$

Rearranging Eq. (3) to solve for the unknown internal energy at State 3 results in the following

$$u_3 = \frac{m_{\rm water}h_{\rm water} + m_2u_2}{m_{\rm water} + m_2} \tag{4}$$

The identity of mass conservation was employed to equate the mass at State 3 to the mass at State 2 and the mass of the injected water. Now that the two properties of internal energy and specific volume are known at state point 3, the thermodynamic state is uniquely determined. Thus the temperature and pressure at the start of the additional power stroke are known.

2.3. Additional power stroke expansion

Similarly to the recompression process, the expansion process from State 3 to State 4 is analyzed and simplified. The temperature

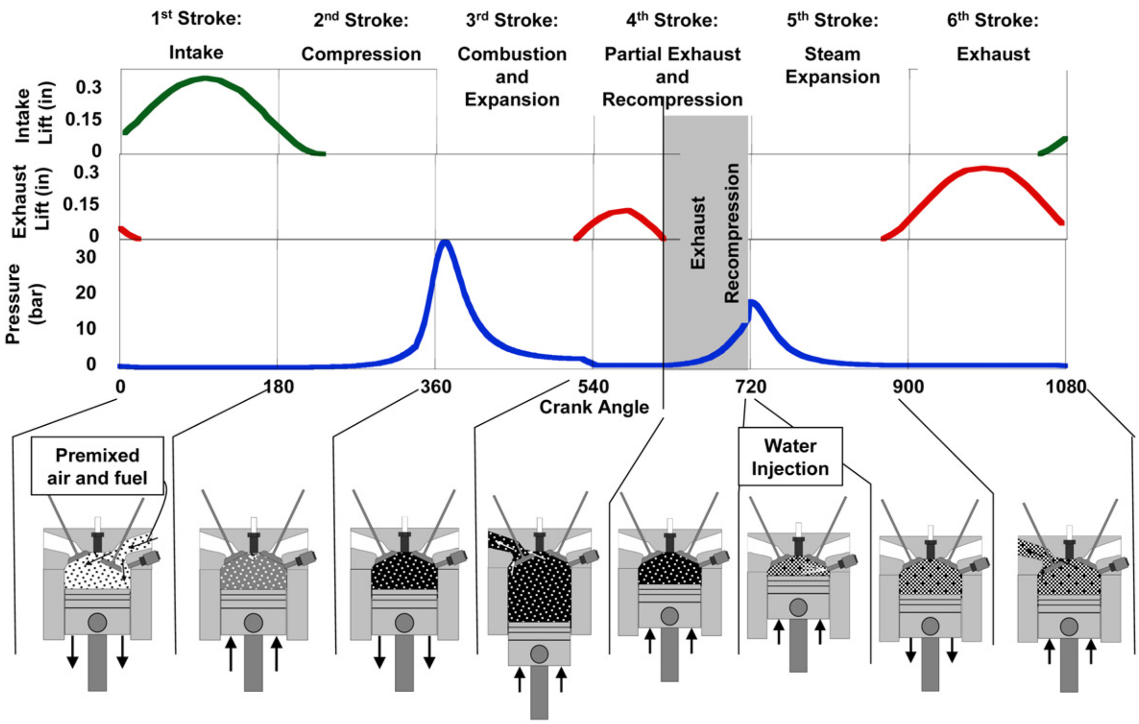

Fig. 6. Example of exhaust valve events and cylinder pressure for the six-stroke cycle.

and pressure at State 3 have been determined from the above analysis. Because there is no mass flow across the combustion chamber control volume during the expansion process and assuming that the recompression process is adiabatic, Eq. (1) reduces to

$$-W_{\mathrm{out}} = E_4 - E_3 \tag{5}$$

Because the mass does not change during this second expansion process to the cylinder volume at bottom center (BC), the thermodynamic state at point 4 is only partially determined with the specific volume being known at TC. An additional assumption that the expansion process is isentropic from State 3 to State 4 yields the additional state property required by the State Postulate to determine completely the thermodynamic properties at State 4. The work output from the expansion process is thus calculated.

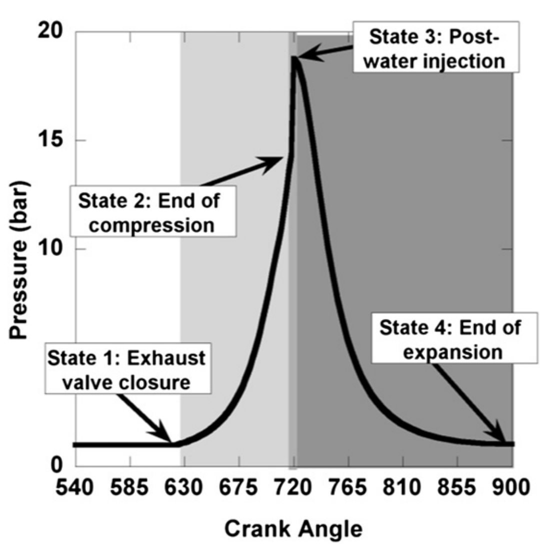

Fig. 7. Pressure trace schematic for exhaust recompression and steam injection showing thermodynamic states.

2.4. Effect of the additional two strokes

Now that the recompression work and the expansion work have been determined, the net work is the expansion work less the recompression work. The net mean effective pressure (MEP) of the early exhaust valve closure and water injection (the fourth and fifth strokes) is then determined by dividing the expansion work of the fifth stroke less the compression work of the fourth stroke by the displacement volume. Although having the units of pressure, the MEP is a measure of the performance of any engine irrespective of size or volumetric displacement. The MEP is a fictitious pressure that if acted on the piston during the entire power stroke would result in the same amount of net work produced in the cycle [9]. Here, the compression work of the exhaust gas residual as shown in Eq. (2) is subtracted from the expansion of the exhaust gas/moisture as shown in Eq. (5) to result in the net work of the additional strokes.

During the calculation of the expansion work using Eq. (5), the combustion gas/water vapor mixture at State 4 was examined for the conditions of pressure, gas temperature, and combustion gas/moisture dew point temperature. If the expansion pressure was less than 1 bar, the calculation was stopped because 1 bar would be the lowest pressure that could exhaust the spent gas to ambient pressure so that the cycle can begin anew. Additionally, if the expansion temperature was less than the dew point temperature, the calculations were also stopped because the condensation of liquid water from the combustion gas/water vapor mixture would cause a decrease in pressure that was not physically modeled which would be parasitic to the expansion work. Condensation during an expansion is generally undesirable because of potential equipment damage due to droplet erosion and also because of the resultant decrease in specific volume. An increase in specific volume results in desirable expansion work.

2.5. Analysis of the combined combustion and water injection events

To complete the analysis of the entire six-stroke cycle, the concept of MEP is further explored. As stated previously, MEP is a fictitious constant pressure acting on the piston during the power stroke that will generate the net work in the cycle as the piston sweeps out the displacement volume in a single expansion. Or in equation form

$$MEP = \frac{W_{net}}{V_{disp}} \tag{6}$$

From inspection of Eq. (6), the fundamental definition of work equal to a force acting through a distance is strictly observed. Thus, MEP term can be thought of as the net work per unit swept volume. MEP for the steam expansion developed here is seen as additional work produced by the engine without an increase in fuel consumption.

Because MEP is formulated as the net work of the cycle, it can be split into the various work terms of each of the strokes. Thus,

$$MEP = \frac{-W_{intake} - W_{compression} + W_{expansion} - W_{exhaust}}{V_{disp}} \tag{7}$$

For a typical four-stroke engine, each of these work terms exists over 180° of crank rotation for a total crank angle of 720° for the complete cycle. For a two-stroke engine, the intake and compression work terms are combined over 180° of crank rotation, and the expansion and exhaust components of the total cycle work are combined over the remaining 180° of crank rotation. For the six-stroke engine proposed here, there are two compression work terms and two expansion work terms each occurring over 180° of crank rotation.

The power and torque of an engine are parameters of particular interest when matching an engine to a desired load. These two parameters can be determined given the MEP and displacement volume of an engine. Because power is simply the product of torque and speed, only an analysis for power is presented here. A formal definition of power equal to work per unit time yields

$$P = \frac{W_{net}N}{n_R} \tag{8}$$

where N represents the engine rotational speed in revolutions per unit time and n_R represents the number of crank revolutions of each power stroke in the cycle. For a two-stroke cycle, n_R is 1 and for a four-stroke cycle, n_R is 2. This n_R term is introduced by Heywood [11] to account for the number of crankshaft revolutions per power stroke.

But what would n_R be for the six-stroke cycle postulated here where there would be one or two power strokes per cycle? If the water injection were turned off while the combustion portion of the cycle continued, n_R would clearly be equal to 3. In this case of zero water injection, the six-stroke engine would develop 2/3 the power of the four stroke at the same engine rotation speed. It would also thus consume 2/3 of the fuel of the equivalent four-stroke engine over the same time period. Setting n_R to 3 is thus done here for consistency with this zero water injection limiting case. Because the amount of MEP for each of the combustion and steam expansions may very well be different, their sum is considered as acting over one-third of the cycle in order to set the value of n_R to 3. By this introduction of n_R, the performance of a particular engine design can be compared with another regardless of the number of power strokes in the cycle.

For the six-stroke cycle, the net work, and thus the net MEP, consists of both a combustion gas expansion event and a steam expansion event. Rewriting Eq. (8) in terms of the MEP and displacement volume results in the following expression for the six-stroke cycle.

$$P = \frac{MEP_{net}V_{disp}N}{3} = \frac{(MEP_{combustion} + MEP_{steam})V_{disp}N}{3} \tag{9}$$

From inspection of Eq. (9), then, if the MEP of the steam expansion (MEP_{steam}) were equal to one-half of the combustion MEP ($MEP_{combustion}$) at the same engine speed, then the power developed by the six-stroke cycle posed here would be equal to the power of the four-stroke combustion cycle alone. In other words, for the power density to remain constant, MEP_{steam} must be equal to one-half $MEP_{combustion}$, which would result in a one-third increase in fuel economy. Thus, the four-stroke engine $MEP_{combustion}$ is 2/3 the six-stroke cycle total MEP at equivalent power and speed conditions when MEP_{steam} is equal to one-half $MEP_{combustion}$.

As will be shown in the following, there are limits as to how much water can be injected and the actual value of the increased efficiency will depend on combustion conditions. In reality, MEP_{steam} will likely be less than one-half of $MEP_{combustion}$, and the resulting efficiency gain will come at some power density penalty. Also, the increased frictional parasitic power consumption of the additional two strokes is not considered in this simplistic analysis, but it is clear that the net power can be approximated at a greatly increased fuel efficiency with the water injection of the additional power stroke included to utilize waste heat of the combustion cycle.

3. Calculated results

As given in the above discussion, there are a number of parameters to be set in order to determine the MEP_{steam} of the resultant early exhaust valve closure with water injection. The initial conditions, assumptions, and constraints for this model are listed in Table 1, and the engine geometry is listed in Table 2.

The initial temperature of 627 °C is higher than the typical temperature shown in Fig. 1. However, the data in Fig. 1 are for an engine equipped with a turbocharger, and the temperature in the current study is a representative temperature for naturally aspirated gasoline engines [11]. The water injection temperature was chosen to be 100 °C because the water can be heated by easily installing a liquid-to-liquid heat exchanger in the engine coolant circuit, which generally has a slightly higher maximum temperature of 105 °C. Thus, this engine cycle is effectively recovering heat from two sources: engine coolant and exhaust gases. The assumption that the injection duration, vaporization, and mixing are all instantaneous, as described in Section 2.2, is not physically

Table 1
Initial conditions, assumptions, and constraints.

Initial conditions	
Initial temperature	627 °C
Initial pressure	1 bar
Gas composition	Complete stoichiometric products of iso-octane (73.4% N_2, 12.5% CO_2, 14.1% H_2O)[a]
Water injection temperature	100 °C
Assumptions	
Water injection duration	Instantaneous at 720 CA
Water vaporization	Instantaneous at 720 CA
Water and air mixing	Instantaneous at 720 CA
Heat transfer to cylinder walls	None
Constraints	
Pressure at 900 CA	≥1 bar
Temperature at 900 CA	≥Dew point

[a] Note that this assumes no humid residual from a previous water injection event.

Table 2
Engine geometry modeled.

Bore (mm)	86
Stroke (mm)	86
Static compression ratio	9.2
Connecting rod length (mm)	147

realistic. However, the assumption that these processes are instantaneous, combined with the assumption of no heat transfer, sets an upper bound on the potential extra work of this engine cycle modification. Finally, the constraints applied to the engine model, as described in Section 2.4, are to ensure that the in-cylinder pressure does not drop below ambient in order to effect removal of the spent gases, and to ensure that liquid does not condense in the cylinder.

Five amounts of water injection at 720 CA are investigated: 0.10, 0.15, 0.20, 0.25, and 0.3 g. These quantities are within the capabilities of the injectors used in the engine given in Table 2. The simulations begin for a 555 CA early exhaust valve closing and then increased to greater exhaust valve closing CA until one of the two constraints listed in Table 1 occurs at the end of the second expansion stroke at 900 CA: either the cylinder pressure decreases during the expansion to 1 bar or the temperature of the cylinder gas/moisture mixture decreases to the dew point temperature.

The modeling results are shown as a function of exhaust valve closing crank angle. Fig. 8 shows the in-cylinder pressure at 900 CA using the timing convention presented in Figs. 4 and 6 (i.e. after the water has been injected and fully expanded). The results are presented as a function of exhaust valve closing crank angle. So the later the crank angle, the higher the position of the piston in the cylinder at the time the exhaust valve closes and the lower the mass of exhaust gases in the cylinder. Fig. 8 shows that for four of the posed water injection masses (0.10, 0.15, 0.20, and 0.25 g), the 1 bar cylinder pressure limitation at 900 CA is reached. The crank angle for closing the exhaust valve for this 1-bar limit is dependent on the amount of water injected. These values are 597 CA for 0.10 g, 611 CA for 0.15 g, 622 CA for 0.20 g, and 635 CA for 0.25 g. As shown in Fig. 8 for the case of 0.30 g of water injected, the pressure at 900 CA is slightly less than 1.2 bar, thus the other limitation at 900 CA of gas expansion temperature less than the dew point temperature halted the simulation.

Fig. 9 shows the in-cylinder temperature of the exhaust gas/moisture mixture when the exhaust valve opens at 900 CA. There are two traces for 0.30 and 0.25 g of injected water that show a temperature slightly less than 100 °C when the exhaust valve opens at 900 CA. For the case of 0.30 g of injected water, the dew point temperature of the mixture of combustion gas and moisture at that point is 98.4 °C, which halted the simulation in order to avoid the presence of undesirable liquid water in the cylinder. For the case of 0.25 g of injected water, the dew point temperature was 93.5 °C because there was less moisture in the exhaust gas mixture in the cylinder. For 0.25 g of water injection as shown in Fig. 8, however, the exhaust pressure was at 1 bar which stopped the simulation just prior to reaching the dew point temperature during the expansion. The remaining parameters of water injection mass of 0.20, 0.15, and 0.10 g show a higher gas temperature at the exhaust valve opening of 900 CA due to the previously shown 1 bar expansion pressure limitation as shown in Fig. 8. The dew point temperature decreased as the mass of water injected decreased, as expected.

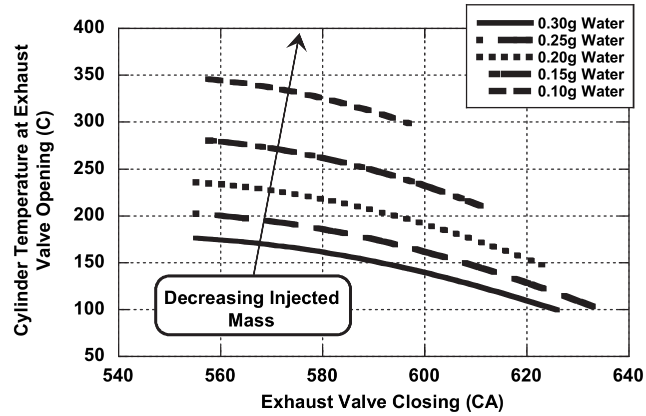

Fig. 9. Cylinder gas temperature at second exhaust valve opening.

Now that the maximum range of conditions of early closing of the exhaust valve subsequent to water mass injection at 720 CA have been determined, MEP_{steam} is calculated by dividing the net work by the displacement volume. The MEP_{steam} for the parameters of injected water mass and early exhaust valve closing is shown in Fig. 10. From inspection of Fig. 10, the MEP_{steam} increases as the amount of water injected increases for any particular value of early exhaust valve closing CA. This is not surprising, as the amount of expansion work from the gas mixture will increase as the amount of mass in the cylinder increases. Interestingly as shown in Fig. 10, the MEP_{steam} increases for any given amount of injected water mass as the early exhaust valve closing occurs at a later crank angle. The reason that the MEP_{steam} increases as the exhaust valve is held open

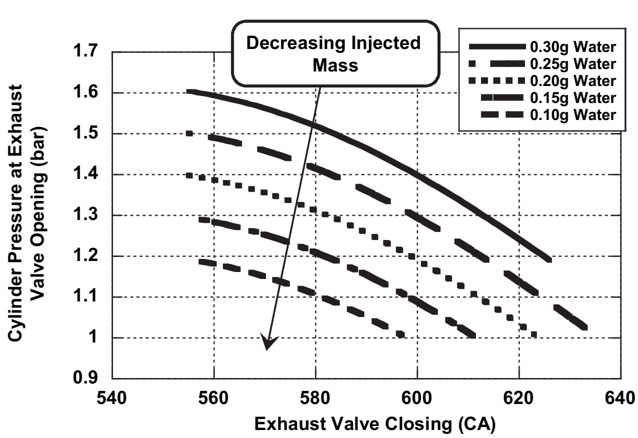

Fig. 8. Exhaust gas cylinder pressure at second opening.

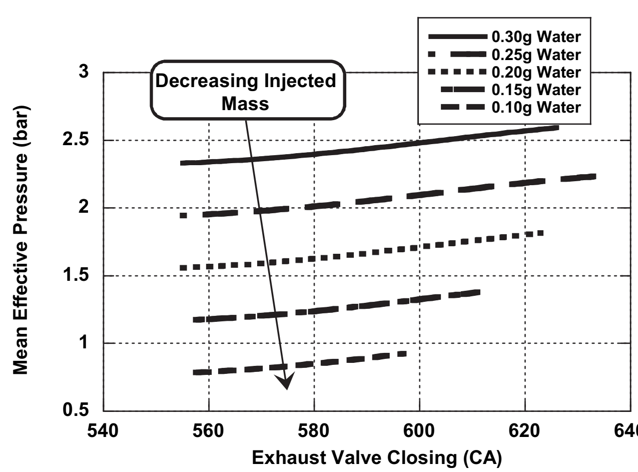

Fig. 10. Mean effective pressure due to second expansion with water injection (MEP_{steam}).

longer is because the recompression work will decrease due to the less exhaust gas residual mass remaining in the cylinder as the exhaust valve stays open longer. From inspection of Fig. 10, any particular value of MEP_{steam} can be obtained from different combinations of water mass injections and exhaust valve closing crank angles.

4. Discussion

As shown in Fig. 10, there is an advantage in delaying the early exhaust valve closing in order to increase MEP_{steam} for any given value of water injection. The value of this exhaust valve closing for maximum MEP_{steam} depends on the limiting conditions of 1 bar or the dew point temperature of the expansion gas/moisture mixture when the exhaust valve opens to discard the spent gas mixture. The range of MEP_{steam} shown in Fig. 10 is from 0.75 to 2.5 bar for one typical gasoline internal combustion engine having geometry as given in Table 1. Typical mean effective pressures of naturally aspirated gasoline engines are up to 10 bar, thus this concept has the potential to show a very significant increase in engine efficiency and fuel economy.

Although the thermodynamic modeling presented here was performed for one set of engine conditions, similar increases in engine output are expected for a wide variety of engine geometries and operating conditions. There is one parameter that is important to the feasibility of this concept, and that is the temperature of the pressurized water injected. As postulated here, the engine coolant heats the injection water to a temperature of approximately 100 °C with a heat exchanger in the coolant circuit. The injection water is moderately pressurized to prevent the water from boiling in the heat exchanger, and pumped to the higher pressure necessary for in-cylinder injection. The sensitivity of the MEP_{steam} to this temperature is noticeable. Calculations with the thermodynamic model for injection water temperatures of 25 °C showed an MEP_{steam} performance decrease of 40% as compared to a water injection temperature of 100 °C and an MEP_{steam} performance increase of 40% if the injection water were heated to 175 °C. Because it should be relatively easy to heat the injection water to 100 °C with a liquid-to-liquid heat exchanger transferring heat from the engine coolant, the 100 °C case is presented here. Although heating the water beyond 100 °C is feasible with either exhaust gas or an external combustion heater, this would require a gas-to-liquid heat exchanger and would require additional engineering considerations beyond the very simple liquid-to-liquid heat exchanger. Thus only the 100 °C water injection temperature is considered for now.

It is also important to note that this is an idealized thermodynamic model and several assumptions were made that may not hold true in a real engine system.

1. It is assumed that water injection, vaporization, and perfect homogeneity are instantaneous. In a real engine system, the vaporization and mixing processes will take a finite time, which could result in a lower power output.
2. Assumptions were made to prevent the in-cylinder temperature from being lower than the dew point and the in-cylinder temperature from being less than 1 bar. These constraints may not be realistic in a real engine when higher temperatures may be needed to prevent condensation throughout the exhaust, and to maintain proper function of exhaust aftertreatment equipment.
3. The thermodynamic modeling does not account for heat transfer between the combustion chamber walls and the cylinder contents. Some heat transfer will occur during the course of the engine cycle, but the thermodynamic modeling shows that there is a sufficient amount of heat in the exhaust gas for the steam cycle without extracting any heat from the walls directly.

5. Conclusions

An ideal thermodynamics model of the exhaust gas compression, water injection at top center, and expansion was used to investigate a modification to recover energy from two waste streams that effectively add two strokes to a common four-stroke internal combustion engine. The additional two strokes require substantial modifications to the exhaust valve operation as well as a manner to inject water directly into the combustion chamber. The hardware necessary to accommodate these modifications to an internal combustion engine is currently available, although significant research is needed to develop the concept. Plans are in place to demonstrate this engine cycle at Oak Ridge National Laboratory (ORNL) on a single-cylinder research engine with fully-variable valve actuation.

Because this injection water is heated by the engine coolant, this six-stroke concept presented here recovers energy from both the engine coolant and combustion exhaust gas. Thus, this concept recovers energy from two waste heat sources of current engine designs and converts heat normally discarded to useable power and work. Provisions may need to be made to keep the exhaust aftertreatment components hot enough to function properly. This concept has the potential of a substantial increase in fuel efficiency over existing conventional internal combustion engines while potentially not decreasing the power density significantly.

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